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Publication date:
2013

Radiant Floor Cooling Combined with Mixing Ventilation in a Residential Room: Thermal Comfort and Ventilation Effectiveness

Michal Krajčík[*#1], Angela Simone[#2], Roberta Tomasi[+#3], and Bjarne W. Olesen[#4]

[*] *Faculty of Civil Engineering, Slovak University of Technology in Bratislava*
Radlinského 11, 813 68 Bratislava, Slovakia
[1] michal.krajcik@stuba.sk

[#] *ICIEE/BYG, Technical University of Denmark*
Nils Koppels Allé, Building 402, 2800 Kgs. Lyngby, Denmark
[2] asi@byg.dtu.dk
[4] bwo@byg.dtu.dk

[+] *Department of Industrial Engineering, University of Padova*
Via Gradenigo, 6/a, 35131 Padova, Italy
[3] rbtomasi08@gmail.com

Abstract

Mixing air ventilation system is one of the main ventilation concepts applied in residential buildings. The effect of combining the mixing ventilation system with the radiant floor heating has been well established, whereas the validation of using the floor for cooling in summer is still in progress. An experimental laboratory study in a simulated residential room with a seated occupant simulated by a thermal manikin was performed in order to evaluate thermal comfort and ventilation effectiveness. Thermal comfort was evaluated by means of vertical air temperature and air velocity profiles and by thermal manikin equivalent temperatures. Contaminant removal effectiveness and air change efficiency were used to characterize the ventilation effectiveness. The vertical air temperature differences that occurred when floor cooling was combined with cold conditioned air supply were well within the limits for comfortable thermal environment recommended by the standards. The cooler supply air mixed well and the effect of the position of air terminal devices was small. When warm unconditioned outside air was supplied by mixing ventilation in combination with the radiant floor cooling, low floor temperature was needed to keep the desired room temperature, followed by increased vertical air temperature differences of about 4 °C for a sitting person, and the ventilation effectiveness was dominated by the position of air terminal devices and the supply air flow.

Keywords: radiant floor cooling; experimental measurements; thermal comfort; ventilation effectiveness; residential building

1. Introduction

Mechanical mixing ventilation system is one of the main ventilation concepts applied in new well insulated and air tight low-energy residential buildings in order to provide fresh air necessary to fulfil the requirements on the fresh air supply.

The mixing ventilation system combined with radiant floor heating is diffusely used in countries where the winter conditions are cold as in Central Europe or in the Scandinavian countries. Such system, often combined with heat recovery, normally guarantees high thermal comfort and at the same time allows efficient utilization of renewable energy sources as solar collector or heat pumps. For this reason it could be practical to use the same system also for cooling in summer.

In summer, the water based surface embedded cooling systems can be combined with natural ventilation or alternatively with a mechanical ventilation system supplying an unconditioned warm outside air in amounts necessary to fulfil ventilation requirements, and thereby limiting the cooling load and allowing better control of the indoor environment. Another possibility is to couple the radiant floor cooling system with conditioned air supplied through the ventilation system at a temperature lower than the room air temperature.

While the effect of combining the mixing ventilation system with the radiant floor heating has been well established, the validation of using the floor for cooling in summer is still in progress. Therefore, thermal environment and air quality in term of ventilation effectiveness were experimentally evaluated in a simulated residential room equipped by a floor cooling system combined with mixing ventilation, at various boundary conditions.

2. Methods

2.1 Experimental Facility

The measurements were carried in the experimental chamber, representing a room in a low energy building during realistic summer conditions in Mediterranean area. Five radiant panels with a total area of 8 m^2 were located on one of the walls to simulate a warm window surface. The room was equipped with a seated thermal manikin and a desk lamp as examples of internal heat sources, a table, a chair, and a hydronic radiant floor cooling. View and layout of the experimental room in tests with mixing ventilation are shown in Fig. 1. A detailed description can be found in [1] and [2].

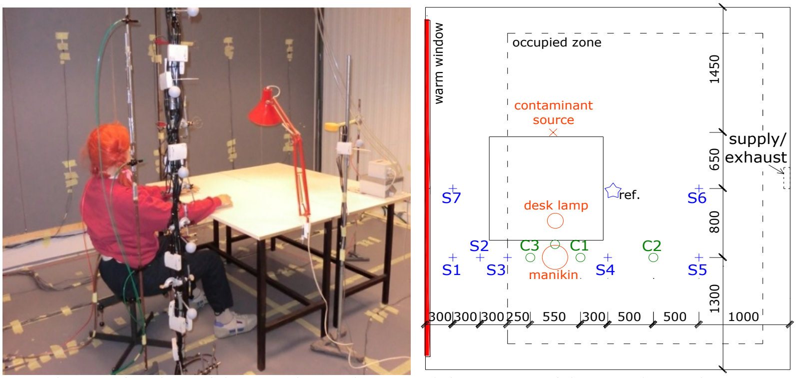

Fig. 1 Left: View of the experimental room. Right: Layout of the experimental room.

2.2 Indoor Environment Quality Indicators

Vertical air temperature and globe temperature profiles, and room surface temperatures were measured in order to describe the thermal environment. Moreover, manikin-based equivalent temperature (t_{eq}) was used to evaluate possible discomfort due to non-uniform thermal environment and local cooling of certain body parts. The manikin-based equivalent temperature can be interpreted as the temperature that a person would sense on various body parts in the actual environment [3]. The manikin was operated in the comfort equation mode, in which the heat supply to particular body segments is maintained equal to the heat loss from that segment, in order to maintain thermal neutrality. The sensible heat loss Q_s can be derived from the power supplied to the body segments. The skin temperatures of the body segments ($t_{sk,i,}$) are then calculated using the following equation:

$$t_{sk,i} = 36.4 - 0.054 \;.\; Q_{s,i} \tag{1}$$

while t_{eq} is calculated for i body segments as:

$$t_{eq,i} = t_{sk,i} - Q_{s,i} / h_{cal,i} \tag{2}$$

where $h_{cal,i}$ is the heat transfer coefficient for a body segment, determined from calibration in a uniform thermal environment. The thermal insulation of the manikin clothing was estimated to 0.6 clo including the chair.

Draught rating (DR) was determined from measured air temperatures, mean air velocities and the standard deviations of air velocities according to [5]:

$$DR = (34 - t_a) \,.\, (v - 0.05)^{0.62} \,.\, (0.37 \,.v \,.Tu + 3.14) \qquad (3)$$

where t_a is the local air temperature in °C, v is the local mean air velocity in m/s and Tu is the local turbulence intensity in %.

Contaminant removal effectiveness (CRE) was used to indicate the ability of the room ventilation system to remove air-borne contaminant released by a simulated passive contaminant source. A perforated table tennis ball covered with a sponge material was used in tests with mixing ventilation as a passive contaminant source with low tracer gas velocity at a short distance from the source. CO_2 was the tracer gas used to simulate the contaminant source. The tracer gas was released at a constant rate on the side of the table opposite the manikin, located 1.1 m above the floor level. When the tracer gas concentration reached steady state, samples were taken in order to describe the CRE at the occupied zone, and from the exhaust air. The CRE was then calculated as:

$$CRE = (c_e - c_s) / (c_i - c_s) \qquad (4)$$

where c_e is the contaminant concentration in the exhaust air, c_i is the mean contaminant concentration in the room and c_s is the contaminant concentration in the supply air. When c_i is the inhaled concentration, ventilation effectiveness represents the personal exposure index [6]. Sample of CO_2 concentration was also taken from breathing zone of the simulated occupant. At complete mixing the concentration at any point in the room is equal to the concentration in the exhaust airflow and the CRE is equal to 1.

Air change efficiency (ACE) is a less direct indicator than the CRE. It characterizes air distribution in the room and can be used when the position of the contaminant source is not specified. The ACE was expressed as a ratio between the local mean age of air at the exhaust and the local mean age of air at a point in the occupied zone. At complete mixing the nominal time constant is the same as the room age of air and the ratio is equal to 1. Values of ACE lower than 1 indicates a short-circuited air flow pattern. The ACE was calculated as:

$$ACE = \tau_n / \bar{\tau}_p \,.\, 100\ (\%) \qquad (5)$$

where τ_n is the local mean age of air at the exhaust, equal to the nominal time constant, and $\bar{\tau}_p$ is the mean age of air at a particular point in the occupied zone. ACE was measured using the step-up method, when the tracer gas was continuously released in the supply air duct at a constant rate. The release of tracer gas started at time t=0 and the increase in the concentration was continuously recorded at a point in the room and in the exhaust airflow. Freon was used as the tracer gas in the air change efficiency measurements. Detailed description of the method can be found in [1], [2] and [7].

2.3 Experimental Cases and Measurement Procedure

The air change of 0.5 h^{-1} representing the minimum requirement for residential buildings according to [8] was used in all studied cases (see Fig. 2 and Table 1), except for case 2, when influence of increased air change (1.0 h^{-1}) was investigated. Case 1 represents floor cooling and supply of warm unconditioned outside air at 30 °C. Case 2 corresponds to case 1, but at the air change of 1.0 h^{-1} in order to investigate the effect of increased air supply. Case 3 represents floor cooling coupled with mixing ventilation supplying cold air at 19 °C. In all cases the supply air temperature was kept constant and floor temperature was adjusted in order to keep the reference temperature equal to 26 °C at 1.1m above the floor (reference is indicated by a blue star in Fig. 1-right). The three cases were performed for two experimental systems, differing in positions of air terminal devices. In System A the supply and exhaust were located in upper part of the wall, whereas in System B the supply was located in upper part of the wall and the extract was just above the floor (See Fig. 1-right and Fig. 2). The measured experimental conditions are summarized in Table 1, along with the results of CRE measurements.

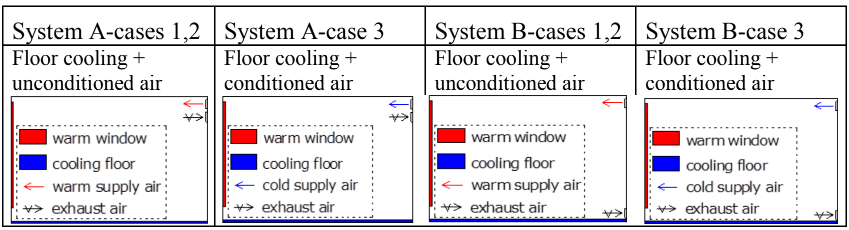

Fig. 2 Setups of the experimental systems tested

Table 1. Experimental conditions and results of contaminant removal effectiveness

System/ case	Nominal ACH (h^{-1})	Heat gain by window (W/m^2)	Internal heat gains (W/m^2)	Supply air temp. minus room temp. (°C)	Room temp. minus window temp. (°C)	Floor temp. minus room temp. (°C)	CRE manikin - Personal exposure index[a]	CRE Avg C1, C2 and C3 at 1.1m	CRE Avg occupied zone[b]
Sys A									
case 1	0.5	25	5.4	4.3	6.6	-6.4	1.15±0.08	1.09	1.10±0.08
case 2	1.0	24	5.3	4.4	6.2	-7.3	0.92±0.04	1.00	0.94±0.19
case 3	0.5	32	5.1	-7.1	8.5	-4.4	1.08±0.00	0.86	0.96±0.11
Sys B									
case 1	0.5	23	5.3	3.6	6.1	-7.1	0.78±0.03	0.75	0.74±0.06
case 2	1.0	21	5.3	3.7	5.4	-7.3	0.97±0.05	1.01	1.00±0.06
case 3	0.5	25	4.9	-7.0	6.6	-2.9	0.93±0.00	0.79	0.83±0.06

a) Mean of three measurements at one position (manikin´s inhalation zone) ± 95 % confidence limit.

b) Mean value from one measurement at nine positions (C1, C2 and C3 at 0.6 m, 1.1m and 1.7m) ± standard deviation.

For the vertical room air temperature/operative temperature profiles and for air velocity profiles measuring positions S3 to S6 in Fig. 1-right were selected in order to characterize the occupied zone, whereas positions S1, S2 and S7 were chosen to describe conditions nearby the window. Air temperatures were measured at 13 heights (from 0.03 m up to 2.3 m) and air velocity profiles were measured at nine heights (from 0.03 m up to 1.7 m) above the floor. In order to assess thermal discomfort caused by draught and vertical air temperature difference, three heights above the floor are recommended by [9]: 0.1 m (ankle level), 1.1 m (head level of a seated person), and 1.7 m (head level of a standing person).

In the CRE measurements the samples were taken from positions C1, C2 and C3 (see Fig. 1-right) at 0.6 m, 1.1 m and 1.7 m above the floor, and from breathing zone of the manikin (personal exposure index). The measurements of ACE were performed in position C1 at 1.1 m and at 1.7 m above the floor, in position C2 at 1.1 m above the floor and in manikin´s breathing zone.

3. Results

3.1 Thermal Environment

The average air temperature profiles in the occupied zone for the two mixing ventilation systems are shown in Fig. 3. Operative temperature was recorded at the same points as the air temperature. The operative temperature profiles were always very close to air temperature profiles and the difference was negligible.

When floor cooling was combined with warm outdoor air supply, the average vertical air temperature difference in the occupied zone exceeded 4 °C and the air velocity was very low, up to 0.07 m/s. For both floor cooling systems the results are similar regardless the nominal air change rate.

For floor cooling combined with cold air supply the average air velocity in the occupied zone was equal to 0.06 m/s and the vertical air temperature difference varied depending on the position of air terminal devices; it was one degree higher when the air was supplied from upper part of the room and exhausted at the floor level. Fig. 4 shows the derived equivalent temperature of each body part for the two experimental systems investigated.

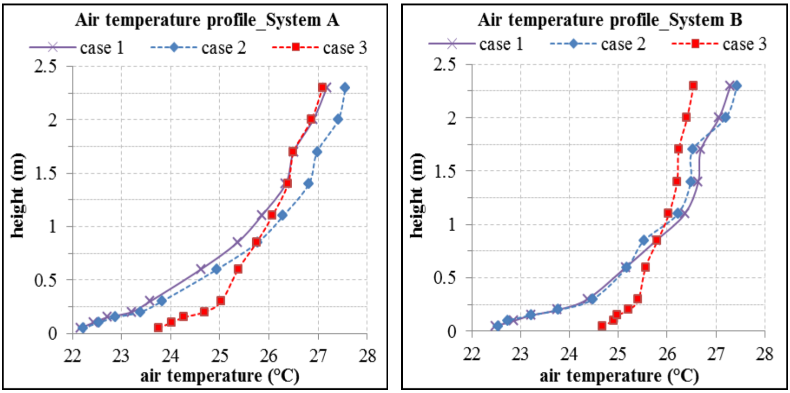

Fig. 3 Average air temperature profiles in the occupied zone

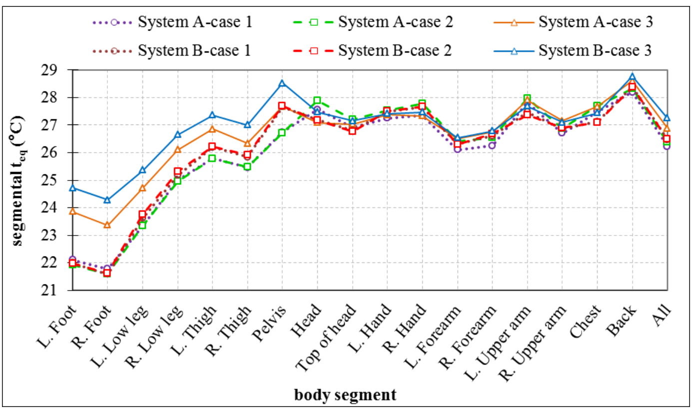

Fig. 4 Equivalent temperature of different body segments of the thermal manikin

3.2 Ventilation Effectiveness

Referring to Table 1, for both mixing ventilation systems the CRE was 0.8 to 1 when floor cooling was combined with cold air supply (case 3). When floor cooling was combined with warm unconditioned air supply (case 1) the position of air terminal devices had an important effect on the CRE at the nominal air change rate as small as 0.5 h^{-1}, while the position of air terminal devices had little effect on the CRE when the nominal air change rate was increased to 1 h^{-1} (case 2).

Results of the air change efficiency measurements are shown for different measurement points in Fig. 5-left for System A and in Fig. 5-right for System B. The values of ACE in the figures are based on mean age of air calculated adding the mean transit time (the time until the molecules of tracer gas reach the measurement point) and the mean presence time [10]. In the analysis the best estimate of mean transit time was made. The error bars in the figures indicate the uncertainty of ACE due to the uncertainty in estimation of the mean transit time. The error bars present absolute values and in the reality, the uncertainty is likely to be less than given by the error bars. The uncertainty of the measurements has been discussed by [2].

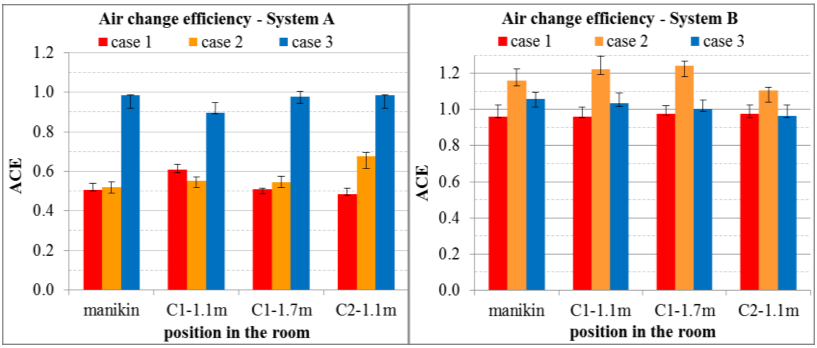

Fig. 5 Air change efficiency (ACE) at different points in the room. Left: System A. Right: System B

4. Discussion

4.1 Thermal Environment

When radiant floor cooling was combined with cold conditioned air supply, the vertical air temperature gradients for a seated person (between 1.1 m and 0.1 m above floor level) were about to 2 °C, corresponding to category "A" of the thermal environment as defined in [5]. The value of t_{eq} for lower body parts (left foot, right foot and left lower leg) some 2-3 °C lower than the t_{eq} on the head should not give rise to problems with thermal discomfort comparing with the comfort criteria for a vertical temperature difference of 3 °C for category "B" [5]. The combination of warm unconditioned air supply with cold floor resulted in increased vertical air temperature differences of about 4 °C, confirmed by the differences between the t_{eq} on the feet and the head of 4 to 5 °C, which represents a thermal environment that is not excellent, but could still be considered within acceptable limits.

Another important aspect of thermal comfort is the low floor temperatures necessary to maintain the reference temperature at 1.1 m above the floor, mainly when unconditioned warm outside air was supplied in the room (cases 1 and 2). In such cases the floor was as low as 19 °C, and in System B - case 2 the heat gain by the window had to be even decreased to reach the desired reference temperature. Even the relatively low floor temperatures meet the requirements on cold floors as defined in [5] and it could be still acceptable e.g. in offices. However, the low floor temperature could cause problems with thermal comfort in residential applications, where the inhabitants are more likely to get in direct contact with the cold floor. This fact may be the main limitation of using floor cooling in residential applications, mainly in warmer environments and thus at higher cooling loads.

In all investigated cases the air velocity was low and the draught ratings always correspond to category "A" of the thermal environment as defined in [5].

4.2 Ventilation Effectiveness

When floor cooling was combined with warm unconditioned air supply (cases 1 and 2) the ventilation effectiveness depended on the position of air terminal devices and on the nominal air change rate (see Table 1 and Fig. 5). In System A with supply and exhaust located in the upper part of the room the ventilation effectiveness was only little dependent on the nominal air change rate. The results of ACE (Fig. 5-left) show significantly short-circuited air flow pattern for both cases, while the CRE is close to 1. In System B with the supply located in the upper part and exhaust at floor level the ventilation effectiveness depended on the nominal air change rate, when both ACE and CRE showed better results for the increased nominal air change rate of 1 h^{-1} (case 2) and the ACE more than 1 indicate good fresh air distribution in the room.

When the cold floor was combined with cold conditioned air supply (case 3), the ACE was always close to 1, regardless of the position of the air terminal devices, indicating good mixing of the cold supply air with the room air.

5. Conclusion

A cold floor during summer presented a potential risk for comfort since it caused high vertical air temperature differences between head and ankle level, in particular at higher cooling loads when warm air entered the room during summer. In such a stratified thermal environment, low floor temperatures were necessary to keep the desired reference temperature. This may be a serious limitation of floor cooling, especially in residential rooms, where the inhabitants are likely to get in direct contact with the cold floor.

When the supply air temperature was higher than the room air temperature, the ventilation effectiveness depended on position of air terminal devices and might vary substantially. Mainly when both supply and exhaust air terminals were located at high level risk of short-circuited air flow pattern was found. When the ventilation air was supplied in the room at a temperature lower than the room air temperature, the ventilation effectiveness was always close to 1, indicating good mixing of the ventilation with the room air and consequently less variation in ventilation effectiveness. From this point of view, supplying the ventilation at or below the room air temperature can be recommended.

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